ANALYSIS OF PASSIVE HEAT REMOVAL SYSTEM THROUGH STEAM GENERATOR

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Abstract: In the current study, the model of Passive Heat Removal System (PHRS) through Steam Generator (SG) was developed for computational fluid dynamics (CFD) analysis. During work there was done three main tasks, which are creating geometrical form of PHRS, meshing of model to achieve a better analysis of thermal-hydraulic phenomena and creating domains for simulation steam condensation in PHRS SG heat exchanger tubes. Analysis through the "ANSYS CFX" software, with two different cases investigated at the steam inlet velocity of 1 m/s and 3 m/s. A thorough description of the steam condensation process in the passive cooling system obtained as a result of the analysis. The rate of the steam condensation in heat exchanger pipeline was assessed.

Keywords: PHRS, CFD, ANSYS, CFX, thermal-hydraulics, NPP

1 INTRODUCTION

Comprehensive experimental and code development research activities have been conducted worldwide to understand thermal-hydraulic phenomena and to establish code predictive capabilities for existing nuclear power reactors. Taking into account that one of the main reasons of Fukushima Daiichi Nuclear power Plant (NPP) accident was loss of the ultimate cooling system, the role of PHRS becomes essential. The main objective of the current work is to assess the possibility and rate of steam condensation in heat exchanger tubes, by which analyze the passive heat removal system‘s heat removal efficiency.

2 DATA COLLECTION FOR PHRS SG

Passive heat removal system is a protective safety system of NPP based on the principle of passive action, designed to provide a long-term heat removal from the reactor core via secondary circuit (Fig. 1)[2]. It performs its functions in all abnormal modes and accidents that bring to passive heat removal from the reactor facility in order to maintain it in safe state, for instance in case of beyond design basis accident involving loss of all power sources. The system consists of four independent trains, one per each steam generator[4]. Steam comes to the PHRS heat exchanger from a pipeline of each SG. Steam is condensed in the heat exchangers by heat removal water tanks. Main objective of current assessment is conceptually to assess possibility and rate of steam condensation in heat exchanger pipeline. Hence for creation of geometrical model only inner/outer diameter in water tank of the system’s heat exchange tube was used, meanwhile other geometrical data was selected arbitrary. System components must withstand seismic impact loads, flooding. System channels are physically separated and totally independent from each other: process parts, control systems, supporting systems, locations of components, pipelines, cables, control elements, etc. are independent, so failure in one channel cannot bring to the failure in another one. This design eliminates dependent failures and common cause failures due to components locations as well as impact of any activities performed on the channel equipment (repair, maintenance) on another channel. Design ensures automatic actuation of the system with passive principle (no need in power supply from external sources or operator interference) [4,5].

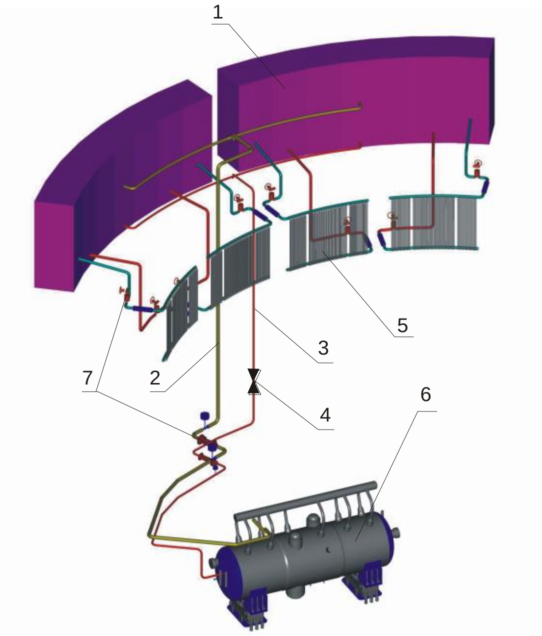

Fig. 1 PHRS SG Components

1 - Emergency heat removal tanks

2 - Steam lines

3 - Condensate lines

4 - PHRS SG valves

5 - Heat exchangers of PHRS HC

6 - SG

7 - Isolating valves

3 GEOMETRY MODEL AND MESHING OF PHRS SG

One heat exchanger tube was modeled by applying geometry described above. Geometrical model was developed using "ANSYS" 16.0 Design Modeler User's Guide [6]. The detailed model was created, which consists of the following bodies: cylindrical heat exchanger tube and surrounding heat exchanger tank.

Parameters	Data
Distance between cold and hot collector	1816 mm
Length of heat exchange tube	2219 mm
Inner diameter of heat exchange tube	12 mm
Outer diameter of heat exchange tube	16 mm

Table 1. Dimensions of the PHRS SG heat exchanger tube

Heat exchanger tubes are submerged in fluid tank with water temperature of 20 °C. For simulation of heat transfer from tube to tank only part of the tank which surrounds the heat exchanger one tube was modeled with size 32x200 mm, height 2100 mm. In accordance to design it is supposed that tubes in two ends are curved approximately by 45 degree to side and bends by 90 degree. The exchanger tube consists of two parts: steam/fluid part and tube's wall. The geometrical model of heat exchanger tube and surrounding heat exchanger tank is presented in Fig. 2.

The partial differential equations that govern fluid flow and heat transfer are not usually amenable to analytical solutions, except for very simple cases. Therefore, in order to analyze fluid flows, flow domains are split into smaller subdomains (grids) that is called meshing. Meshing model was developed using "ANSYS" 16.0 Meshing User's Guide [7].

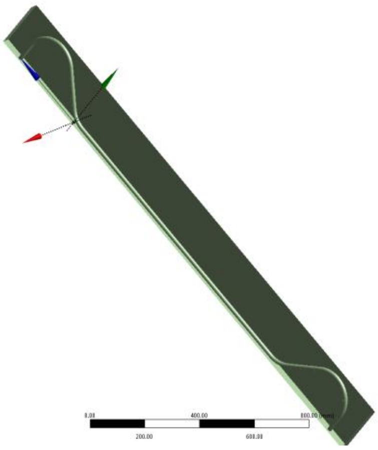

Fig. 2 Geometrical model of PHRS SG

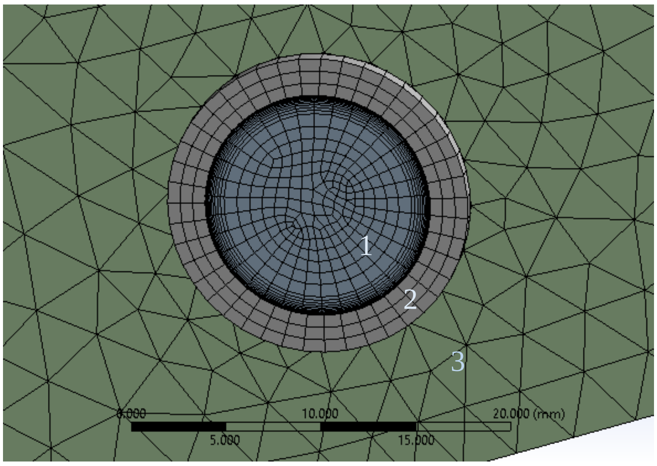

Fig. 3 Meshing of PHRS SG
1 - Steam/fluid part, 2 - Tube's wall,
3 - Water tank

4 CFD MODEL OF PHRS SG

During implementation of current task it was observed from "ANSYS" 16 CFX-Solver Modeling Guide [8] applicability of "Equilibrium Phase Change" model of condensation. A multicomponent flow consists of two or more gases mixed on a microscopic level. For such a flow, single velocity, temperature and turbulence field are calculated. The amount of the different gases, which can vary in space and time, determine the properties of the fluid. In a multiphase flow, the fluids are mixed on a macroscopic length scale, and separate velocity and temperature fields can be solved for each fluid. The equilibrium phase change model assumes local thermodynamic equilibrium between two phases (e.g. liquid water and vapor). Condensation occurs as soon as the saturation temperature for the given static pressure has been obtained for the water vapor in the flow. Since local thermodynamic equilibrium is assumed, a single temperature field can be solved for the mixture. The flow is considered homogeneous, thus a single velocity field is solved for the mixture, reducing the computational power needed to obtain a solution.

For the model mentioned above three domains were created for simulation of condensation phenomena. First domain was created for simulation of water/steam inside the heat exchanger tube. Material properties of the coolant were set according to the material library [11]. The buoyancy model was used with the buoyancy reference pressure equal to 782 kg/m3 (which satisfies density of water at 7 MPa and 286 °C). The heat transfer mode was set to "Thermal Energy", which took into account the transport of enthalpy and included kinetic energy effects. The turbulence model was set to the Shear Stress Transport model (CFX-Solver Modeling Guide Release 16, 2016 [9]). Second domain was created to model heat exchanger tube wall. Material of tube wall was selected as steel. Third domain was created to model PHRS SG tank. Material properties of the coolant were set according to material library [11]. The buoyancy model was used with the buoyancy reference pressure equal to 1000 kg/m3 (which satisfies density of water at pressure of 0.1 MPa and temperature 20 °C).

Based on known diameter of tube and assumption that steam injected to tube will be fully condensed at the outlet, it is possible to evaluate velocity of steam at the inlet using the following equation

$$v = \frac{W}{(L)} \frac{1}{\rho_v S} = 2.33\ m/s \tag{1}$$

Where W is heat removal by one tube, L is the latent heat, ρ is density of steam and S is cross section area of tube. To assess possibility and rate of steam condensation in heat exchanger pipeline two variants with different inlet velocity were analyzed: 1 m/s and 3 m/s, which represent lower and upper possible operational boundaries of the system [2,4,5].

Parameters	Boundary type	Option 1	Option 2
Velocity of steam (m/s)	INLET	1	3
Mass fraction of steam at inlet		0.99	0.99
Temperature (°C)		286	286
Pressure (MPa)	OUTLET	7	7
Temperature (°C)		286	286
Temperature of tube's outer wall (°C)	Wall	20	

Table 2. Initial and boundary conditions

5 RESULTS OF ANALYSIS

a) Option 1. Inlet velocity 1 m/s

The vapor injected to the tube begins to condense due to heat transfer from steam to cold wall. The formed liquid film continues to flow through the tube until reaching the curved part of the tube. At the center of the bend, the liquid film detaches from the tube wall and pours down to the opposite wall due to tangential velocity. Flow of the fluid continues after the bend until reaching the second bend of the tube and pours down to the opposite wall again.

As the liquid film detaches from the tube wall and pours down, it diverges into many streams that partially evaporate due to being heated up by the vapor. This process results in reduction of liquid void fractions at the detachment location (Fig. 4). After that the flow of the fluid continues in vertical part of the tube. Due to condensation along the tube the volumetric fraction of liquid continues to increase. In vertical part of the tube the liquid film starts to flows downward due to gravity. Due to steam condensation additional new condensate added to liquid film as it flows downward. At the left half sector of tube wall, water film thickness is higher than in right half sector. This is conditioned by the liquid film which was detached from the above and pours down to the left sector of wall. The same process as described above takes place also for the lower part of the tube.

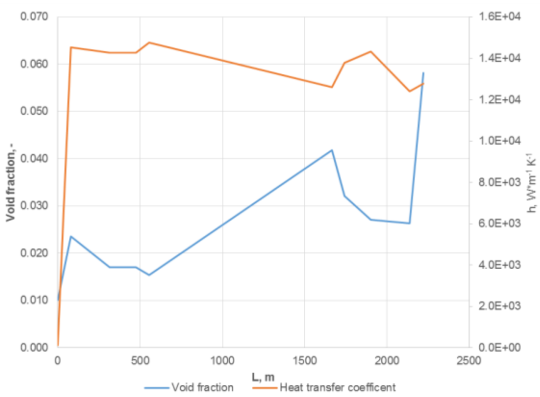

Fig. 4 Average heat transfer coefficients of water liquid film and void fraction (option 1)

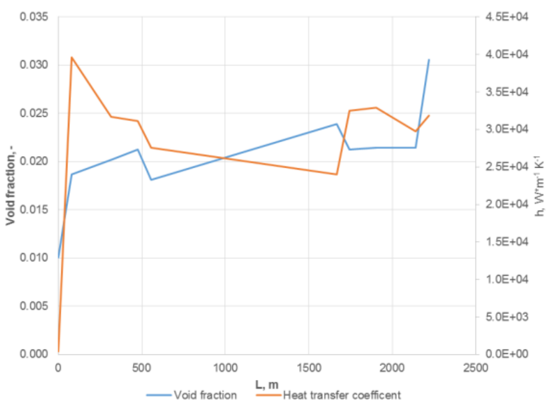

Fig. 5 Average heat transfer coefficients of water liquid film and void fraction (option 2)

The liquid condensation rate is calculated as the difference of liquid mass flow rate at the inlet and outlet and is equal to 2.3 g/s. Accordingly the amount of heat removed by one tube will be equal to

$$W = GL = 3.46\,kWt \tag{2}$$

where G is a condensate liquid mass rate and L is a latent heat.

b) Option 2. Inlet velocity 3 m/s

The condensation process in tube in case the injected steam velocity is equal to 3 m/s is similar as in case for 1 m/s. Main difference is total amount of condensate liquid which in case of 3 m/s steam velocity is equal to 0.7 g/s. Consequently, the amount of heat removed by one tube will be equal to 1.02 kWt. Relatively small amount of condensate liquid in comparison with the first case is conditioned by high speed of injected steam which passes through tube during 0.7 s and doesn't manages to cool so as in option 1.

CONCLUSION

In the current study, the model of Passive Heat Removal System through Steam Generator was developed for CFD analysis. Three domains were created for simulation of steam condensation in PHRS SG heat exchanger tubes. Results of the assessment of lower and upper possible operational boundaries showed that in case of steam inlet velocity 1 m/s the total heat removal capacity will be equal to 3.4 kWt. However, the results of the analysis with higher steam injection velocity (3 m/s) showed that condensation rate will decrease up to 1 kWt.

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